Rotary Centrifugal and Viscous Pumps

FIELD OF INVENTION

The present invention relates generally to micro-scale pumps, millimeter-scale pumps, and macro-scale pumps.

BACKGROUND

The area of microfluidics is developing with many new sensors, separation devices, drug delivery systems, and other small-scale and micro-scale fluidic devices. For many of these devices there is a need to circulate or move fluid through macro-scale and micro-scale channels. A variety of micropumps are available to meet this need, and generally to fulfill specific applications. For example, such micropumps include membrane pumps, electrohydrodynamic (EHD) pumps, electrokinetic (EK) pumps, rotary pumps, peristaltic pumps, bubble based pumps, etc. Non-mechanical pumps, like the electrohydrodynamic and electrokinetic pumps, do not have moving parts, which increases reliability. Such devices, however, are generally limited by low flow rate and pressure rise capabilities, the applications of the pump, the working fluids that can be pumped, and high supply voltage requirements. Mechanical pumps, like rotary pumps, peristaltic pumps, and membrane pumps, have a wide variety of possible working fluids and applications. Such mechanical micro-pumps (like rotary micropumps), however, are believed to be feasible only when they are greater than a certain size.

Miniature rotary pumps have been proposed. For example, see Hainan, C., Ahaoying, Z., Yong, L., Xiongying, Y., and Yihua, Y., 1997, "A novel centrifugal miniature pump and its medical application," Proceedings of the 1997 International Symposium On Micromechatronics and Human Science, IEEE, pp. 115-117. Such rotary pumps have an impeller to produce pumping. The behavior and performance of macro-scale rotary pumps are well established and well known. The dynamics, performance, and efficiency of rotary pumps, however, change as the size of the pump is altered. For example, secondary flows, and the losses associated with them, become more important as macro-scale impeller passage size decreases. As size decreases further, surface (viscous) forces become more significant, and can dominate the performance of the pump.

One important dimension of many rotary pumps is the gap distance between the top of the blades, and the pump housing. As the gap increases, there is more leakage across the blades, and the overall hydraulic efficiency of the pump decreases. If the gap

is too small or zero, the blades can be damaged by contacting the pump housing. On a macro-scale this "gap problem" is generally insignificant, but on a micro-scale, or a millimeter-scale, the gap between the top of the blades and the pump housing can be about the same as the height of the impeller blades. Thus, in a miniature rotary pump, the blades should be close to the pump housing to minimize cross-blade leakage and losses between the blades and the housing, but such closeness risks damage or friction losses due to contact.

The Hainan pump is reported to have a maximum flow rate of 100 ml/min and a maximum pressure rise of 10 kPa.

Another rotary micropump has an integrated magnetic motor with a stator that also acts as the impeller. See Ahn C. H. and Allen, M. G., 1995, "Fluid micropumps based on rotary magnetic actuators," Proceedings of IEEE Micro Electro Mechanical Systems 1995, IEEE, ASME, pp. 408-418. The pump has radial inflow and radial outflow. Such a pump, however, is believed to have lower performance, i.e. lower flow rate and lower pressure rise. For example, the Ahn pump is reported to have a maximum flow rate of 24 μl/min, and a maximum theoretical pressure rise of 100 Pa.

Another type of pump has been proposed that includes one spinning disk spinning adjacent a spiral channel concentric with and spiraling around the rotational axis. An inlet is formed perpendicular to the disk at the inside of the spiral near a center of the disk. An outlet is formed perpendicular to the disk at an outside of the spiral near an outer edge of the disk. See Kilani, M., Galambos, P., Haik, Y., and Chen, C. J., 2003, "Design and Analysis of a Surface Micromachined Spiral-Channel Viscous Pump," ASME Journal of Fluids Engineering, 125 (2), pp. 339-344. It is believed that the Kilani pump may be difficult to manufacture.

Another type of pump has been proposed that includes a shaft with eccentric rotation nearer one side of a tube than another. See Sen, M., Wajerski, D., and Gad-El-Hak, M., 1996, "A Novel Pump for MEMS Applications, " ASME Journal of Fluids Engineering, 118, (3), pp. 624-627. The Sen pump utilizes the outer surface of the shaft. The Sen pump has limited flow rates and pressure rise capabilities.

Another type of pump has been proposed that utilizes a stack of spinning disks.

SUMMARY

It has been recognized that it would be advantageous to develop a millimeter-scale or micro-scale pump which imposes fluid motion and pressure rise by means of viscous and inertial forces.

Briefly, and in general terms, the invention is directed to a pump device with means for both 1) transferring rotational momentum to the fluid by viscous forces, and 2) imparting centrifugal forces to the fluid.

In one embodiment of the invention set forth above, the pump can include a wiper extending partially across at least one rotatable disc surface of at least one rotatable disc.

In another embodiment of the invention set forth above, the pump can include a rotatable shaft with a hollow cavity extending from one end to at least one slot extending radially from the hollow cavity.

BRIEF DESCRIPTION OF THE DRAWINGS

Additional features and advantages of the invention will be apparent from the detailed description which follows, taken in conjunction with the accompanying drawings, which together illustrate, by way of example, features of the invention; and, wherein:

FIG. 1a is a schematic, partial cross-sectional side view of a pump in accordance with an embodiment of the present invention;

FIG. 1b is a schematic, partial cross-sectional side view of another pump in accordance with another embodiment of the present invention;

FIG. 1c is an exploded perspective view of the pump of FIG. 1b;

FIG. 1d is a graph of flow rates versus rotational speed for the pumps of FIGs. 1a and 1b, with a disc diameter of 2.38mm and a gap height of 103 μm;

FIG. 2a is a partial, internal top view of the pump of FIG. 1a;

FIG. 2b is a partial, internal top view of another pump in accordance with another embodiment of the present invention;

FIG. 2c is a partial, internal top view of another pump in accordance with another embodiment of the present invention;

FIG. 2d is a partial, internal top view of another pump in accordance with another embodiment of the present invention;

FIG. 2e is a partial, internal top view of the pump of FIG. 1b;

FIGs. 3a-3f are perspective views of discs of the pumps of FIGs. 1a and 1b in accordance with embodiments of the present invention;

FIGs. 3g-3i are partial perspective internal views of wipers of the pumps of FIGs. 1a and 1b in accordance with embodiments of the present invention;

FIG. 4a is a schematic, partial cross-sectional side view of another pump in accordance with another embodiment of the present invention;

FIG. 4b is a partial, cross-sectional perspective view of the pump of FIG. 4a;

FIG. 4c is a cross-sectional side view of the pump of FIG. 4a;

FIG. 4d is a graph of pressure head versus volumetric flow rate for various experimental pumps with a constant impeller rotational rate of 9,240 rpm, and with the data points from left to right for each impeller configuration obtained with outlet tubing diameters of 0.254 mm, 0.508 mm, and 1.397 mm, respectively;

FIG. 5a is a partial, cross-sectional perspective view of a shaft or impeller of the pump of FIG. 4a;

FIGs. 5b-5f are partial, cross-sectional perspective views of shafts or impellers for the pump of FIG. 4a in accordance with embodiments of the present invention;

FIG. 5g is a partial top view of a shaft or impeller for the pump of FIG. 4a in accordance with an embodiment of the present invention;

FIG. 6a is a schematic, internal top view of the pump of FIG. 4a showing a volute configuration; and

FIGs. 6b-6d are schematic, internal top views of volute configurations for the pump of FIG. 4a in accordance with embodiments of the present invention.

DETAILED DESCRIPTION OF EXAMPLE EMBODIMENT(S)

As illustrated in the Figures, various pumps are shown as exemplary implementations in accordance with the invention. The size of the pumps can be scaled up to have dimensions on the order of millimeters, inches, or feet, or can be scaled down to have the dimensions on the order of micro-meters. The pumps utilize both viscous and centrifugal forces to provide pumping action. The pumps have a rotational or spinning element that transfers rotational momentum to the fluid by viscous forces, and imparts centrifugal forces to the fluid. At very small scales and for some operating conditions and embodiments, the centrifugal forces may be insignificant compared to the viscous forces. Fields that can benefit from the use of such micro-scale and millimeter-scale pumps

include electronics cooling, implantable medical devices, μ-TAS (micro Total Analysis Systems), micro-fluid dispensing systems, transport of biomedical fluids, drug delivery, chemical analysis systems, etc. Fields that can benefit from the use of such millimeter-scale and macro-scale pumps include blood pumping, ventricle assist devices, etc.

The pumps described herein can be described as two types. The first type is a spinning disc type pump, indicated generally at 10a and 10b, and shown in FIGs. 1a-3h, that utilizes a spinning disc as the rotational element with a wiper extending partially across the rotatable disc surface. The second type is a rotary shaft type pump, indicated generally at 10c and shown in FIGs. 4a-6d, that utilizes a rotatable shaft as the rotational element with a hollow cavity extending from one end to slots extending radially from the hollow cavity.

As illustrated in FIGs. 1a-3h, a spinning disc type pump 10a is shown in an example implementation in accordance with the invention. The pump can have a pump housing 14 with an inlet 18, an outlet 22, and a fluid passage 26 through which a fluid can pass. The pump housing can be formed by, or can include, various components of the pump. Similarly, the fluid passage can be formed by, or can include, various components of the pump. In addition, the pumps can include an inlet channel 30 between the inlet 18 and the spinning disc, and an outlet channel 34 disposed between the spinning disc and the outlet 22.

As described above, the pump 10a includes a wiper 44 extending partially across a rotatable disc surface 48 of a rotatable disc 52. The disc surface can be the end of a shaft while the disc can be the shaft. The disc surface 48 contacts the fluid in the fluid passage 26, and transfers momentum to the fluid through viscous forces, which move the fluid through the pump or fluid passage. In addition, centrifugal forces are present in the pump due to the rotation of the fluid through the pump, which aids in pumping by forcing the fluid outward toward the outlet channel 34. Thus, the fluid can pass through the inlet channel 30 to the spinning or rotatable disc surface 48, which transfers momentum to the fluid through viscous forces, which moves the fluid through the outlet channel 34.

The wiper 44 is disposed adjacent, or on, the disc surface 48, and extends across a portion of the disc surface 48. The wiper 44 "wipes" the fluid from the disc surface 48 and/or directs the fluid from the disc surface 48 towards the outlet channel 34. The wiper 44 can include a leading edge 56 which guides the fluid, and an opposite trailing edge 60. The leading edge 56 of the wiper 44 can help define or form a portion of the fluid passage

26 and the outlet channel 34. Similarly, the trailing edge 60 of the wiper 44 can help define or form a portion of the fluid passage 26 and the inlet channel 30.

The pump 10a can include an opposing wall 64 that opposes the wiper 44 or the leading and trailing edges 56 and 60. The opposing wall 64 can be spaced-apart from the wiper 44, and can include a portion positioned along a perimeter or outer circumference of the disc surface 48 to maximize the surface area of the disc surface in contact with the fluid. The wiper 44 (or leading and trailing edges 56 and 60), the opposing wall 64 and the disc 52 (or disc surface 48) together define at least a portion of the fluid passage 26 (and the inlet and outlet channels 30 and 34).

The disc surface 48, inlet channel 30, outlet channel 34, and fluid passage 26 can be aligned, or oriented parallel to one another, to reduce interference in the flow of the fluid, reduce pressure and flow losses, and increase efficiency. Alternatively, the inlet and outlet channels can have different configurations or orientations.

The wiper 44 can extend to an axis of rotation 68 of the disc 52, thus extending across half the disc surface 48, as shown in FIG. 2a. The leading and trailing edges 56 and 60 of the wiper 44 can be oriented orthogonal to one another, again as shown in FIG. 2a. Similarly, the opposing wall 64 can have orthogonal portions, or can form a right angle. Thus, the pump 10a can be configured with the inlet and outlet 18 and 22, or the inlet and outlet channels 30 and 34, orthogonal to one another.

The disc surface 48 can be configured to accommodate desired flow characteristics, and/or to optimize efficiency. For example, the disc surface 48 can be configured to enhance momentum transfer from the disc surface to the fluid. Referring to FIG. 3a, the disc surface 48b can include one or more channels or elongated indentations 72 extending into the disc surface 48b and oriented radially. For example, the disc surface 48b can include four channels or indentations 72 extending from a center of the disc to a perimeter of the disc, and oriented at equal angles (right or ninety-degree angles) with respect to one another. The channels or indentations 72 can be linear or straight, as shown. In addition, the channels or indentations 72 can have rectilinear side walls, or rectilinear cross-sections, as shown. It is of course understood that more or fewer channels can be used; that the channels can be arcuate or curved; and that the walls or cross-sections of the channels can be curved or angled. More channels might increase momentum transfer and increase turbulence or mixing. Curved channels might be used to

impart desired flow characteristics. Curved or angled walls of the channels might be used to facilitate laminar or turbulent flow.

Referring to FIG. 3b, the disc surface 48c can include one or more ridges or elongated protrusions 76 extending from the disc surface 48c and oriented radially. For example, the disc surface 48c can include four ridges or protrusions 76 extending from a center of the disc to a perimeter of the disc, and oriented at equal angles (right or ninety-degree angles) with respect to one another. The ridges or protrusions 76 can be linear or straight, as shown. In addition, the ridges or protrusions 76 can have rectilinear side walls, or rectilinear cross-sections, as shown. It is of course understood that more or fewer ridges can be used; that the ridges can be arcuate or curved; and that the walls or cross-sections of the ridges can be curved or angled. More ridges might increase momentum transfer and increase turbulence or mixing. Curved ridges might be used to impart desired flow characterizes. Curved or angled walls of the ridges might be used to facilitate laminar or turbulent flow, or mixing.

Referring to FIG. 3c, a plurality of dimples 80 can extend into the disc surface 48d. The plurality of dimples 80 can be positioned in a linear array that can be oriented radially. Furthermore, the disc surface 48d can include a plurality of linear arrays extending radially from near a center of the disc to a perimeter of the disc, and oriented at equal angles with respect to one another. The dimples 80 can be of uniform diameter and/or depth, and can have curvilinear walls or cross-sections. As stated above, the dimples can enhance momentum transfer. It is of course understood that the dimples can have non-uniform sizes (i.e. different diameters and/or depths); can have rectilinear or angled walls or cross-sections; or oriented in curvilinear arrays. Non-uniform dimples or curvilinear arrays might impart desired flow characteristics. Rectilinear or angled walls or cross-sections can be used to facilitate laminar or turbulent flow or mixing.

Referring to FIG. 3d, the disc surface 48e can include one or more arcuate blades 84 extending from the disc surface, and oriented in a spiral about the axis of rotation 68 of the disc. The blades 84 can have a proximal end near the center of the disc or near the axis of rotation, and spiral (i.e. extend radially and circumferentially) to an outer circumference or perimeter of the disc. The arcuate blades can help push the fluid towards the outlet channel 34. The blades 84 define corresponding arcuate channels 86 therebetween. It is of course understood that the disc surface 48e can be described as

having one or more arcuate channels 86 extending into the surface and oriented in a spiral.

The above described features can enhance momentum transfer from the disc surface to the fluid. Fluid velocity variations will occur as the fluid moves past the above described features, resulting in eddies and local turbulence, which will increase the diffusion of momentum normal to the disc surface. This will increase pump efficiency and performance. In addition, the features may help push the fluid toward the outlet channel.

The disc surface 48 also can include patterned or random surface roughness to enhance momentum transfer. For example, increased surface roughness can be used to enhance momentum transfer, turbulent flow and/or mixing, while decreased surface roughness can be used to enhance more laminar flow. The surface roughness may be patterned to obtain specific flow characteristics at various radial positions on the disc surface.

Referring to FIGs. 3e and 3f, the disc can have a non-planar disc surface. The disc surface 48f can have a cone or conical shape with a longitudinal axis collinear with the axis of rotation 68 of the disc, as shown in FIG. 3e. Similarly, the disc surface 48g can have a curved or dome shape with a longitudinal axis collinear with the axis of rotation 68 of the disc, as shown in FIG. 3f. The cone or domed shape of the disc surface 48f or 48g creates a greater thickness or width of the fluid passage near the perimeter or circumference of the disc, and less thickness or narrower width of the fluid passage near the axis of rotation. The area between the disc and top of the housing is referred to as the gap. The non-planar shape of the disc surface alters a gap height with respect to radial position. This change in gap height can be used to eliminate backflow in the pump, which can increase pump efficiency and performance. The wiper 44 can be configured to match the inclined or curved surface of the disc.

The wiper 44 can also be configured to accommodate desired flow characteristics, and/or to optimize efficiency. Referring to FIG. 3g, the wiper 44b can include one or more ridges or protrusions 90 extending from the leading and/or trailing edge 56 and/or 60. In addition, the wiper 44b can include one or more channels or indentations 94 extending into the leading and/or trailing edge 56 and/or 60. The ridges or channels 90 or 94 in the edges 56 and 60 of the wiper can increase mixing of the fluid. Fluid can become trapped, and can circulate in and out of the channels, thus enhancing mixing. The ridges

or channels can be rectilinear, as shown. It is of course understood that the ridge or channel can be curvilinear or angled to facilitate a more laminar flow. The ridges or channels can extend through or across a thickness of the wiper, or through the gap, as shown.

In addition, the wiper 44 can be configured to reduce friction with the disc 52. Referring to FIG. 3h, the wiper 44c can include one or more ridges or protrusions 98 extending from the wiper 44 in a direction away from, or towards, the disc 52 or disc surface 48. The ridges or protrusions 98 can extend along the leading and trailing edges 56 and 60. Similarly, the wiper 44c can include one or more channels or indentations 102 extending into the wiper. The channels or indentations can extend into the wiper opposite the disc surface. The ridges 98 and channels 102 can reduce friction between the disc surface 48 and the wiper 44c, such as by reducing the area of the wiper 44c in contact with the disc 52 or disc surface 48. Reducing friction can reduce the power requirements to rotate the disc. (The ridges 98 and channels 102 shown in FIG. 3h are configured to face a second disc, described below, but the wiper can be flipped-over to face the first disc 52, the configuration shown in FIG. 3h being for clarity.)

The wiper 44 can be configured to facilitate mixing, and increase performance of the wiping action. Referring to FIG. 2b, the wiper 44d, or the leading and trailing edges 56d and 60d, can extend across the disc 52 or disc surface 48 a distance greater than a radius of the disc surface, or can extend across more than half the diameter of the disc. Such a configuration can decrease the recirculation near the center of the disc surface. Alternatively, referring to FIG. 2c, the wiper 44e, or the leading and trailing edges 56e and 60e, can extend across the disc or disc surface a distance less than a radius of the disc surface, or can extend across less than half of the diameter of the disc. Such a configuration can increase the recirculation near the center of the disc surface.

Referring to FIG. 2d, the wiper 44f can have leading and/or trailing edges 56f and 60f with a curvature in a plane parallel with the rotatable disc surface 48 or orthogonal to an axis of rotation 68 of the rotatable disc surface. The curvature can increase the efficiency of the wiper 44f in "wiping" the fluid from the disc surface 48. In addition, the leading and trailing edges 56f and 60f can extend at an acute angle with respect to one another. Decreasing the angle between the edges 56f and 60f can increase the exposed disc surface 48, thus increasing pump performance. Alternatively, the edges can be oriented at an obtuse angle depending on the application of the pump.

Referring again to FIG. 2b, the leading and trailing edges 56d and 60d can have an intersection or tip 106 with a convex curvature therebetween. The convex curvature can reduce recirculation and mixing, but can increase pumping performance. Alternatively, referring to FIG. 2c, the leading and trailing edges 56e and 60e can have an intersection 110 with a concave curvature. The concave curvature can increase recirculation and mixing.

The wiper 44 can also be configured to affect the flow rate, pressure rise and/or mixing characteristics. Referring to FIGs. 2b and 2d, the wiper 44d and 44f can be positioned to extend across the axis of rotation 68 of the disc 52, and across over half of the disc. Referring to FIG. 2c, the wiper 44e can be positioned to extend aside from, or away from, the axis of rotation 68 of the disc 52, and across less than half of the disc.

In addition, the leading and trailing edge of the wiper can be angled to change the efficiency of the wiper to "wipe" the fluid from the disc surface. The edge 56 or 60 of the wiper 44 can be orthogonal or perpendicular to the disc surface 48, as shown in FIG. 3h. Alternatively, the edge 56d or 60d of the wiper 44d can have an acute or obtuse angle with respect to the disc surface, as shown in FIG. 3i. In addition, the edge of the wiper can be curved in a plane orthogonal to the disc surface.

As shown and described above, the wiper can be positioned and oriented symmetrically with respect to the disc surface. Alternatively, the wiper and disc surface can have an asymmetrical configuration.

In addition, the inlet and outlet channels, or the inlet and outlet, of the pump can be configured to optimize performance or to accommodate particular applications. For example, the inlet and outlet channels 30 and 34 can be oriented parallel with the disc surface 48 and orthogonal to one another, as shown in FIG. 1a. As another example, the inlet and outlet channels 30b and 34b can be oriented at an obtuse angle with respect to one another to maximize the disc surface 48, as shown in FIG. 2d. Alternatively, the inlet and outlet channels can be oriented perpendicularly or inclined with respect to the disc surface to suit particular applications.

Referring to FIG. 1b, another pump 10b is shown that is similar in many respects to the pump 10a of FIG. 1a, and described above. The pump 10b can include the wiper 44 disposed between a pair of spaced-apart discs 52b and 52c rotatably disposed in the pump housing. The discs 52b and 52c include opposing disc surfaces 48h and 48i. The wiper 44 can be disposed between the discs 52b and 52c and can wipe or direct the fluid

from both disc surfaces 48h and 48i. The discs 52b and 52c can be interconnected by a common shaft 120 so that they rotate together at the same speed, and can be turned by the same motor. Alternatively, the discs 52b and 52c can be rotated separately, or at different speeds, to enhance mixing of the fluid. The discs 52b and 52c, or the disc surfaces 48h and 48i, can be similar to facilitate manufacturing, facilitate laminar flow and minimize recirculation. For example, the discs or disc surfaces can have the same diameter and the same surface roughness, can be planar, and can be parallel to one another. Alternatively, the discs or disc surfaces can have different diameters, different surface roughness, can be non-planar, and/or can be non-parallel to increase recirculation or mixing, as discussed above.

During testing of the spinning disc type pumps 10a and 10b, three discrete regions of a pumping area, fluid passage or disc surface were observed, as shown in FIG. 2e. In a first region (region 1), near a leading edge of the disc and the trailing edge 60 of the wiper 44, the spinning disc 52 draws fluid away from the trailing edge of the wiper, creating a low pressure region. The pressure forces, due to the pressure gradient, force fluid from the inlet channel 30 into the pumping area, and along the trailing edge of the wiper. The fluid then is forced tangentially through a second region (region 2) due to the viscous forces applied to the fluid by the spinning disc and the diffusion of momentum. The fluid tangential velocity decreases as the fluid approaches the leading edge 56 of the wiper 44 near the outlet in a third region (region 3), which results in a high pressure region at the trailing edge of the wiper. The high pressure will force the fluid radially outward toward the outlet channel 34. Due to the variation of tangential velocity with radial position, the region of highest pressure may be located at a radial position between an outer disc radius and a coupling shaft radius, resulting in a fluid flow from the region of highest pressure toward the coupling shaft. This movement of fluid results in a backflow at smaller radii of the pumping area. Centrifugal force also forces the fluid from the pumping area to the outlet channel. The fluid flow through these regions was visualized in the double-disc pump 10b using a dye.

The height or thickness of the fluid passage through the pump (also height of the wiper) is referred to as the gap height of the pump. The volume of fluid between the two discs or between the single disc and upper pump housing is referred to as the pumping area. Experimental pumps have been created with a disc diameter of 2.381 mm, and a gap height of 103 μm. The maximum flow rates obtained were 0.74 ml/min and 2.1

ml/min for the single-disc and double-disc pumps, respectively, for a rotational speed of 5000 rpm.

The advantages of the above-described micro-scale pumps compared to other micropumps include a wide range of possible flow rates, simplicity, constant flow, planar structure, well controlled flow rate, and possible mixing characteristics.

The discs 52-52c described above can be formed by the end of a shaft. Referring to FIG. 1c, the shafts can be captured in bearings 116b and 116c, which also form a seal along the shafts to reduce leakage.

An exploded view of an experimental double-disc pump can be seen in Figure 1c. There are five main fabricated components of the experimental pump: 1) the disc(s) 52b and 52c, 2) top housing 122, 3) bottom housing 124, 4) bearings 116b and 116c, and 5) wiper/channel insert 128. Fabrication of the disc(s) 52b and 52c can be realized using precision machining techniques. A lathe can be used to obtain the desired outside diameter of the disc shaft, and to create the disc surface. The diameter of the experimental disc(s) for flow rate testing was 2.381 mm, and the diameter of the experimental discs for flow visualization was 6.35 mm. For the double-disc pump, a hole can be bored into one of the disc surfaces, in which the coupling shaft 120 (FIG. 1b) can be press-fit. The other disc can be machined on the lathe to the desired outside diameter and then the coupling shaft can be machined on the end, such that the step between the disc shaft diameter and coupling shaft diameter forms the other disc surface. The coupling shaft can be press-fit into the bored hole in one disc until the desired gap height is obtained. The disc of the single-disc pump can be made from type 304 stainless steel. The discs for the double-disc pump used for flow rate testing were made from brass. The bottom and upper discs for the double-disc pump used for flow visualization were made from aluminum and acrylic respectively. The acrylic disc was clear and allowed the flow to be visualized through the disc. The above-described parts can be integrated together to produce the same features with fewer parts. For example, the wiper/channel insert can be integrated into the top housing. As another example, the bearing material can be the same material as the top housing, such that the top housing and top bearing are the same piece.

For experimentation, the top housing 122 was made from acrylic to allow the flow passing into and through the pump to be visualized. The top housing can have an inlet and outlet passage that connects the inlet and outlet tubing to the inlet and outlet channels of the pumps. The top housing 122 for the double-disc pump contains the upper bearing

116c. The bottom housing 124 for the pumps can be made from aluminum, and contains the lower bearing 116b.

The bearings 116b and 116c can be made from Torlon, a strong self-lubricating plastic. The upper and lower bearings can press-fit into the top and bottom housing respectively. Torlon is used because it has a low coefficient of friction, and can be easily machined. The outer diameter of the bearings was 12.7 mm, and the inner diameter was the diameter of the disc shaft. A recess that is 1 mm deep was cut into the bottom bearing under the inlet and outlet channel. The larger hydraulic diameter of the inlet and outlet channels due to this recess reduces the speed and pressure losses of the fluid as it travels through the inlet and outlet channels, and minimizes losses as the fluid turns from the inlet tubing to the inlet channel, and from the outlet channel to the outlet tubing.

The wiper/channel insert 128 can be machined from brass shim stock using a CNC milling machine. The brass shim stock used for the flow rate testing and flow visualization testing was 103 μm tall and 78 μm tall respectively. The width of the inlet and outlet channels machined into the brass shim stock was half the disc diameter (channel widths of 1.191 mm, and 3.175 mm). The wiper/channel insert 128 can be captured between the top and bottom housings 122 and 124 and a seal is created by tightening bolts that clamp the top and bottom housing together.

The pump can be assembled by connecting the disc pump shaft to the motor coupling shaft on the motor. The disc surface can be adjusted so that the disc surface is flush with the top surface of the bottom housing. The wiper/channel insert can then be positioned and secured in place. Then for the double-disc pump, the coupling shaft with the upper disc can be press-fit into the lower disc until the upper disc surface contacts the upper wiper surface. The top housing can then be secured, and tightened to minimize leakage. The inlet and outlet tubes can then be connected to the top housing. The air can be bled from the pump.

As discussed above, the pumps can be powered, and the discs can be rotated, by a motor. The motor can be selected based on size and power requirements of the desired application. For experimentation, the disc pumps were powered by an externally mounted Maxon EC32 number 118891, brushless DC motor that is 32 mm in diameter, with an 80 Watt power rating. The maximum speed is 25,000 rpm, with a stall torque of 0.35 N-m. The brushless motor was controlled by an Advanced Motion Controls power amplifier (Model #BE12A6). The power amplifier has a DC supply voltage of 40V, a

peak current of 12 Amps, and continuous current rating of 6 Amps. A negative feedback controller was employed to maintain constant speed for any variation in torque. The speed was controlled by adjusting a 15-turn potentiometer. The speed range was 100-13,200 rpm. The motor controller determines the rotational speed from the signal from an optical encoder attached to the motor shaft. This apparatus produced a voltage signal that is proportional to speed. The voltage was measured using a Keithley 131 Digital Multimeter.

Referring to FIG. 1d, experimental flow rates for the experimental single-disc and double-disc pumps 10a and 10b are shown for rotational speeds ranging from 1000 rpm to 5000 rpm, disc diameter of 2.381 mm, and gap height of 103 μm. The experimental results show that the flow rate increases nearly linearly with rotational speed for the single-disc and double-disc pumps. The maximum flow rates for the single-disc pump and double-disc pump are 0.74 ml/min and 2.1 ml/min respectively for a rotational speed of 5000 rpm. Note that the flow rate will increase with greater rotational speed, and larger disc diameter. The flow rate will decrease with lower rotational speeds, smaller disc diameters, and smaller gap heights. The flow direction can also be reversed by changing the rotational direction of the disc(s).

The advantages of these micro-scale pumps compared to other micropumps include a wide range of possible flow rates, simplicity, constant flow, planar structure, well-controlled flow rate, and possible mixing characteristics. The experimental flow rate shows a nearly linear relationship between flow rate and rotational speed. The fluid pumping direction can be reversed by changing the rotational direction of the disc(s). The design of the disc pumps is simple, and can potentially be fabricated using microfabrication technology due to the planar structure of the pump. The disc pumps are actuated with an external motor, but a microfabricated disc pump can potentially be actuated with a micro-electrostatic comb drive. The movement of the dye through the pumping area indicates the presence of a recirculating flow near the coupling shaft, and the movement of dye due to pressure gradients, viscous forces, and centrifugal forces. The movement of the dye also indicates a significant increase in the surface area of the dye volume, which will increase mixing through the pump area and outlet channel.

As illustrated in FIGs. 4a-6d, a rotatable shaft type pump 10c is shown in an example implementation in accordance with the invention. As described above, the pump 10c includes a rotatable shaft 150 with a hollow cavity 154 extending from one end to one

or more slots 158 extending radially from the hollow cavity. The hollow cavity 154 can be a bore extending axially into one end of the shaft so that a longitudinal axis of the hollow cavity 154 is concentric with a longitudinal axis of the shaft 150, or an axis of rotation 162 of the shaft. The hollow cavity 154 can have an inner surface 166 that contacts the fluid and helps define a fluid passage 26c. The slots 158 can extend from an outer surface of the shaft 150 to the inner surface 166 of the hollow cavity 154, and can be positioned intermediate the ends of the shaft. The slots 158 can define impellers or impeller blades 170 that are contained within the circumference or perimeter of the shaft.

The rotatable shaft 150 or inner surface 166 transfers momentum to the fluid through viscous forces, and provides a "pre-swirl" to the fluid that aids in overall pumping by reducing sudden acceleration of the fluid at the impellers. In addition, centrifugal forces act on the fluid between the impeller blades 170, or in the slots 158, forcing the fluid outward, and thus creating a pumping effect. Thus, the fluid can pass into the hollow cavity 154 to the inner surface 166, which transfers momentum to the fluid through viscous forces, and through the slots 158, which apply centrifugal forces to the fluid.

A pair of bearings 174 and 178 can carrying the shaft 150, and can be disposed on opposite sides of the slots 158 and impeller blades 170. An intermediate member 182 can be disposed between the pair of bearings 174 and 178, and can define a volute 186 around the shaft 150 at the slots 158.

The bearings 174 and 178 and intermediate member 182 can be contained in, or can help form, a housing 14c. As described above, the housing 14c or pump 10c can include an inlet 18c and an outlet 22c, and an inlet channel 30c extending from the inlet to the shaft and an outlet channel 34c extending from the shaft, slots or volute to the outlet. The hollow cavity 154 can form part of the inlet channel 30c. The inlet and outlet channels 30c and 34c can be oriented parallel with the shaft 150 or axis of rotation 162. In addition, the housing or pump can include an exit plenum 190 disposed proximate the volute 186 so that the fluid passes through the volute and into the exit plenum. Alternatively, the volute 186 can open to an outlet.

The hollow cavity 154 or bore can have a straight, constant diameter and can be oriented concentric with the axis of rotation 162, as shown in FIGs. 4a and 5a. In addition, the shaft 150 can have a tapered inlet 194 to reduce fluid losses. The hollow cavity can also be configured to reduce fluid losses. For example, the hollow cavity can

include a tapered bore 150b with a straight wall in cross-section along the axis of rotation 162 of the shaft, as shown in FIG. 5e. The bore can be inwardly or outwardly tapered. Alternatively, the hollow cavity can include a tapered bore with a curved wall in cross-section along the axis of rotation of the shaft.

In addition, the hollow cavity 154 or inner surface 166 can be configured to increase momentum transfer from the inner surface to the fluid, and to help direct or push the fluid through the hollow cavity. For example, the hollow cavity or inner surface can include one or more ridges 200 extending from the inner surface 166 and oriented parallel with the axis of rotation 162 of the shaft, as shown in FIG. 5c. Similarly, the hollow cavity or inner surface can include one or more channels 204 extending into the inner surface and oriented parallel with the axis of rotation of the shaft, as shown in FIG. 5c. As another example, the hollow cavity or inner surface can include one or more spiral blades 208 extending from the inner surface, or one or more spiral channels 210 extending into the inner surface, as shown in FIG. 5d. The spiral ridges or channels can extend around the inner surface and along the length of the inner surface. The ridges or channels can have rectilinear cross-sections, as shown, to increase momentum transfer. It is of course understood that the ridges or channels can have curved or inclined cross-sections to reduce turbulence or mixing. Furthermore, the inner surface can have random or patterned surface roughness, as described above.

The slots 158 can be configured to increase pumping performance. For example, a plurality of slots 158 can be disposed laterally adjacent one another and disposed circumferentially around the shaft, as shown in FIG. 5a. As another example, a plurality of slots 158 can be disposed longitudinally adjacent one another along a length of the shaft, as shown in FIG. 5b. In addition, the plurality of slots can be disposed circumferentially and longitudinally, again as shown in FIG. 5b. More or larger slots can increase the volume available for fluid flow, can decrease fluid losses, and can increase pump performance. Similarly, the plurality of impeller blades 170 can be laterally adjacent one another and disposed circumferentially around the shaft, as shown in FIG. 5a, and/or longitudinally adjacent one another along a length of the shaft, as shown in FIG. 5b.

The slots 158 or blades 170 can be configured to improve pump performance. For example, the slots or blades can have a leading or trailing edge 214 that is flat and oriented parallel with the axis of rotation of the shaft, as shown in FIG. 5f. As another

example, the leading or trailing edges 218 can have a concave curvature, as shown in FIG. 5a. Alternatively, the leading or trailing edge can have a convex curvature.

The slots or blades 220 can be arcuate, or have a curvature, in a plane perpendicular to the axis of rotation, as shown in FIG. 5g. The curvature can be oriented to curve forward or backward with respect to the direction of rotation of the shaft. For example, an experimental backward curved 4-blade impeller was fabricated with β1 and β2 angles of 80 and 85 degrees respectively. As another example, an experimental forward curved 4-blade impeller was fabricated with β1 and β2 angles of 144 and 114 degrees respectively. In addition, the blades 220 can be narrower than the slots to increase the area through which the fluid can pass.

In addition, the slots 158 or blades 170 can be configured to enhance heat transport near the surface. For example, the leading or trailing edges can have a similar or different surface roughness.

Furthermore, the impeller blades can be configured to extend into the hollow interior.

Referring to FIG. 5f, an end surface 230 of the hollow cavity 154 proximate the slots 158 can have a protrusion 234 to guide flow to the at least one slots. The protrusion 234 can have a concave curvature, as shown. In addition, the walls of the hollow cavity 154 can have a similar curvature leading into the slots.

Referring to FIG. 6a, the volute 186 can be configured to facilitate flow from the shaft or impeller blades. The volute 186 can have a volute wall 240 with a concave curvature surrounding a portion of the shaft, and extending in an open configuration to allow the flow to exit the volute into the exit plenum 190, or outlet. Thus, there is an unobstructed channel between the volute and the plenum, and the volute wall extends outward from the shaft without narrowing.

An experimental pump was built with a rotatable shaft 150 with a 2.38 mm outer diameter and a hollow cavity with a 1.17 mm inner diameter. The rotatable shaft 150 was mounted using bearings 174 and 178 located above and below the exit plenum 190, which are mounted in the pump housing 14c. The volute 186 and an outlet channel are then located in the region between the upper and lower bearings 174 and 178 and pump housing 14c. There can be a continuous channel, from the inlet tubing, through the pump housing and top of the upper bearing to the inlet of the shaft. Inside the channel through the upper bearing, the fluid flow transitions from a non-rotating bearing wall to the inside

of a rotating shaft. The bearing forms a seal for the spinning shaft, which reduces the leakage from the impeller outlet to the shaft inlet. The side walls of the volute and part of the outlet channel for an experimental pump were formed by a piece of machined brass shim stock that is 416 μm tall. This volute and outlet channel can be aligned with the slot ports of the shaft. With this construction, when the shaft spins, centrifugal forces from the spinning impeller shaft force fluid flow through the shaft inlet, through the interior of the shaft, through the slots, out through the slot ports, into the volute, and then into and through the outlet channel. The volute design shown in FIG. 6a has been found to minimize the effects of surface forces, and is referred to as an open volute design. The open volute design is characterized by a large "open" channel from the impeller to the exit plenum, or outlet. Alternative volute designs are shown in FIGs. 6b-6d.

The volute 186 can efficiently direct fluid toward the outlet channel. The volute designs employed in macroscale pumps, where fluid motion is induced by inertial forces, are different from the design employed here. This design difference is because flow from the impeller exit and within the volute is significantly influenced by both inertial forces and surface forces. The present open volute design increases the width and maximizes the hydraulic diameter thereby decreasing the average fluid velocity and velocity gradients, which also reduces viscous losses.

The pump 10c advantageously includes the impeller integrated into the body of the shaft, instead of on the top of the shaft. This design is simple, and allows it to be easily manufactured at low cost. In addition, the problems of tip blade clearance and flow leakage around the blade tips are not present. As stated above, an important dimension of many macro-scale centrifugal pumps is the gap distance between the top of the blades, and the pump housing. As the gap increases, there is more leakage across blades, and the overall hydraulic efficiency of the pump decreases. If the gap is too small or zero, the blades can be damaged by contacting the pump housing. On a macro-scale this "gap problem" is generally insignificant, but on a micro-scale, or a millimeter-scale, the gap between the top of the blades and the pump housing can be about the same as the height of the impeller blades.

The design of the rotary shaft pump (RSP) eliminates this "gap problem." The shaft or impeller can be constructed by boring a hole in the end of a shaft, and then cutting slots in the side of the shaft at the bottom of the bored hole. Thus, the material between the slots acts as the blades of the impeller, and the slots form passages between

the bored interior and outer shaft surface. The gap at the top of the blades is zero, because the top of the blades also connects to the shaft. The hollow interior of the shaft transfers momentum to the passing fluid through viscous forces. This "pre-swirl" can be significant on a millimeter-scale or micrometer-scale, when the ratio of the hollow interior length to diameter is large, and the circumferential wall velocity is greater than the average axial fluid velocity. The "pre-swirl" aids in the overall fluid pumping by reducing the sudden acceleration of the fluid at the inner blade tip.

An experimental rotary shaft pump was fabricated with five main parts: 1) the impeller or shaft 150, 2) top housing, 3) bottom housing, 4) bearings 174 and 178, and 5) volute 186. Fabrication of the impeller 150 can be realized using precision machining techniques. A lathe can be used to obtain the desired outside diameter of the shaft, and to bore the hole in the end of the shaft. The cone shape at the inlet of the shaft can be made using a center drill. Slots can then be cut into the side of the shaft using a milling machine and an indexing tool. The impeller can be made from 304 stainless steel. Five impeller designs were constructed for testing: 1) radial 2-blade, 2) radial 4-blade, 3) radial 6-blade, 4) backward curved 4-blade, and 5) forward curved 4-blade. The blades were evenly spaced around the circumference of the shaft. The blades on the impellers were evenly spaced, and were 0.38 mm tall (the slots are 0.38 mm tall). The backward curved 4-blade impeller has $\beta 1$ and $\beta 2$ angles of 80 and 85 degrees respectively. The forward curved 4-blade impeller has $\beta 1$ and $\beta 2$ angles of 144 and 114 degrees respectively. The above described parts can be integrated together to produce the same features with fewer parts. For example, the volute can be integrated into the top housing. As another example, the bearing material can be the same material as the top housing, such that the top housing and top bearing are the same piece.

The top housing 202 was made from acrylic to allow the flow passing into and through the pump to be visualized. The top housing contains the upper bearing 174 and can have an inlet channel and outlet channel. The ends of clear plastic tubes can be press-fit into the holes. The bottom housing 206 was made from aluminum, and contains the lower bearing 178. The bottom housing can also have a recess that forms the exit plenum 190. The expansion of the fluid channel as it enters the exit plenum slows the fluid, and minimizes losses as the fluid turns to the outlet channel.

The bearings were made from Torlon, a strong self-lubricating plastic. The upper and lower bearings were press-fit into the top and bottom housing respectively. Torlon is

used because it has a low coefficient of friction, and can be easily machined. The outer diameter of the bearings was 6.35 mm, and the inner diameter was 2.38 mm.

The volute 186 was machined from brass shim stock using a CNC milling machine. Each piece of brass shim stock was 104 μm thick. Thus the height of the volute can be changed by stacking pieces of brass shim stock. For all of the present tests, a height of 0.416 mm was used because it is just larger than the height of the slots located in the shaft.

The pump can be assembled by connecting the impeller shaft 150 to the coupling shaft on the motor. The volute 186 can then be positioned and secured in place. The top housing 202 can then be secured, and tightened to minimize leakage. The inlet and outlet tubes can then be connected to the top housing. The air can be bled from the system.

As described above, the shaft can be operatively coupled to a motor. The experimental pumps were powered by an externally mounted Maxon EC32 number 118891, brushless DC motor that is 32 mm in diameter, with an 80 Watt power rating. The maximum speed is 25,000 rpm, with a stall torque of 0.35 N-m. The maximum testing speed was limited by the maximum speed of the optical encoder (14,000 rpm) attached to the motor shaft. The brushless motor was controlled by an Advanced Motion Controls power amplifier (Model #BE12A6). The power amplifier has a DC supply voltage of 40V, a peak current of 12 Amps, and continuous current rating of 6 amps. A negative feedback controller was employed to maintain constant speed for any variation in torque. The speed was controlled by adjusting a 15-turn potentiometer. The speed range is 100-13,200 rpm. The motor controller determines the rotational speed from the signal from an optical encoder attached to the motor shaft. This apparatus produces a voltage signal that is proportional to speed. The voltage was measured using a Keithley 131 Digital Multimeter.

Experimental pumps were tested with experimental impeller designs include radial 2-blade, radial 4-blade, radial 6-blade, backward-curved 4-blade, and forward curved 4-blade arrangements. The fluid flow for each impeller design was throttled using outlet tubing diameters of 0.254 mm, 0.508 mm, and 1.397 mm of the same length.

Referring to FIG. 4d, variations of pressure head with volumetric flow rate are shown for the pumps, all for an impeller rotational speed of 9,240 rpm. The data was obtained by throttling the flow in the rotary shaft pump experiments by changing the inside diameter of the outlet tubing. The data for each pump arrangement (from left to

right) correspond to outlet tubing inner diameters (D) of 0.254 mm, 0.508 mm, and 1.397 mm, respectively. For each impeller configuration (operating at constant impeller rotational speed), FIG. 4d shows that increasing the exit tube diameter gives a higher volumetric flow rate through the pump, which, for the impellers tested, gives lower magnitudes of pressure head. Qualitative trends of FIG. 4d data for each impeller configuration (at constant impeller rotational speed) are similar for $Q > 2$ ml/minute. Here, different data sets have similar slopes, sometimes with different ranges of pump head. At lower volumetric flow rates, data sets for different impeller configurations sometimes show completely different slopes, as well as different quantitative trends. Overall, data trends shown in FIG. 4d are similar to ones also observed at impeller rotational speeds of 1,320 rpm and 13,200 rpm. Pressure head magnitudes produced by the impeller generally decrease as the flow rate is increased (as outlet tubing diameter increases) for a constant rotational speed. This is associated with a decrease of hydraulic efficiency, which is related to overall impeller behavior, and to increased hydrodynamic losses and increased slip as flow rate increases. The nearly parallel nature of the pressure rise and flow rate trends for the different impeller blade designs indicates that these overall slopes are independent of impeller blade design. The impeller blade design thus appears to affect the magnitude of the product of hydraulic efficiency and slip factor, but not overall slopes of the pressure - flow rate data.

The above-described experimental pumps produced flow rates up to 64.9 ml/min. This value is larger than many other micropumps of similar size, which generally reach flow rates only up to about 16 ml/min.

The centrifugal pumps described above advantageously have high efficiency and reduce the possibility of breaking micro-blades. For example, the centrifugal pumps can have 80% hydraulic efficiency at 2 ml/min flow rate. Traditional centrifugal pumps can require impeller blades to be positioned close to the stationary top of the pumping chamber, but with a gap that lowers the efficiency. Reducing the gap increases the risk that the blades impact the top of the pumping chamber, breaking the blades or causing other damage. The centrifugal pumps described above, however, essentially have a gap of zero between blades and the top because the top directly above the blades is also part of the shaft. In addition, the centrifugal pumps described above can be easily manufactured.

A larger-scale experimental pump was tested. The pump impeller was 25.4 mm in diameter. The volute was machined into the top housing with a depth of 3.81 mm, such that the volute and top housing were one piece. The pump was tested with an outlet tubing inner diameter and length of 9.525 mm and 300 mm respectively. The rotational speeds tested were 1500-3300 rpm. The maximum flow rate achieved was 3200 ml/min, with a maximum pressure of 7.6 kPa.

While the foregoing examples are illustrative of the principles of the present invention in one or more particular applications, it will be apparent to those of ordinary skill in the art that numerous modifications in form, usage and details of implementation can be made without the exercise of inventive faculty, and without departing from the principles and concepts of the invention. Accordingly, it is not intended that the invention be limited, except as by the claims set forth below.